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# APPLICATION OF PHYSICAL PARAMETER IDENTIFICATION TO FINITE-ELEMENT MODELS*

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**Abstract**

The time domain parameter identification method described in Volume I is applied to TRW's Large Space Structure Truss Experiment. Only control sensors and actuators are employed in the test procedure. The fit of the linear structural model to the test data is improved by more than an order of magnitude using a physically reasonable parameter set. The electro-magnetic control actuators are found to contribute significant damping due to a combination of eddy current and back EMF* effects. Uncertainties in both estimated physical parameters and modal behavior variables are given.

# 1 Summary

The availability of transient test data from TRW's Large Space Structure Truss Experiment (LSSTE) allowed the parameter identification procedure to be verified against actual hardware. Various member stiffness and mass properties, structural damping, magnetic damping in the control actuators, and actuator gains were the parameters which were adjusted to better match the model to reality. Using the approximation concepts approach, an orders of magnitude improvement in computational efficiency was obtained over previous efforts.

The use of Prony's method to fit exponentially damped sinusoids to the test data allowed visual verification of the linear damping assumption. This revealed that the primary source of damping in the control actuators for moderately large motions was due to magnetic hysteresis, and not friction. The large amount of damping available from electro-magnetic control actuators suggests their use for suppression of high frequency vibrations outside an active control system's bandwidth.

# 2 Test Sequence

The LSSTE is shown in Figure 1. The structure is basically a frame, with a fairly rigid top plate made of honeycomb supported by four thin columns. The primary modes are two lateral modes and a torsional mode of the top plate which involve primarily bending flexibility of the columns. The diagonal members carry no axial loads other than friction and damping forces, and control forces when the active vibration suppression system is turned on. The control system is designed to actively damp out vibrations induced by a pair of random disturbance generators located on the top plate. In order to employ a strong control algorithm, an accurate knowledge of plant behavior is necessary, hence, the need for parameter identification. It was the objective of this test–analysis correlation effort to employ only control sensors and actuators in order to simulate an on–orbit procedure.

A series of five tests were performed using various combinations of the four control actuators to apply initial forces and then suddenly release the load. Two lateral

* EMF (electro–motive force)

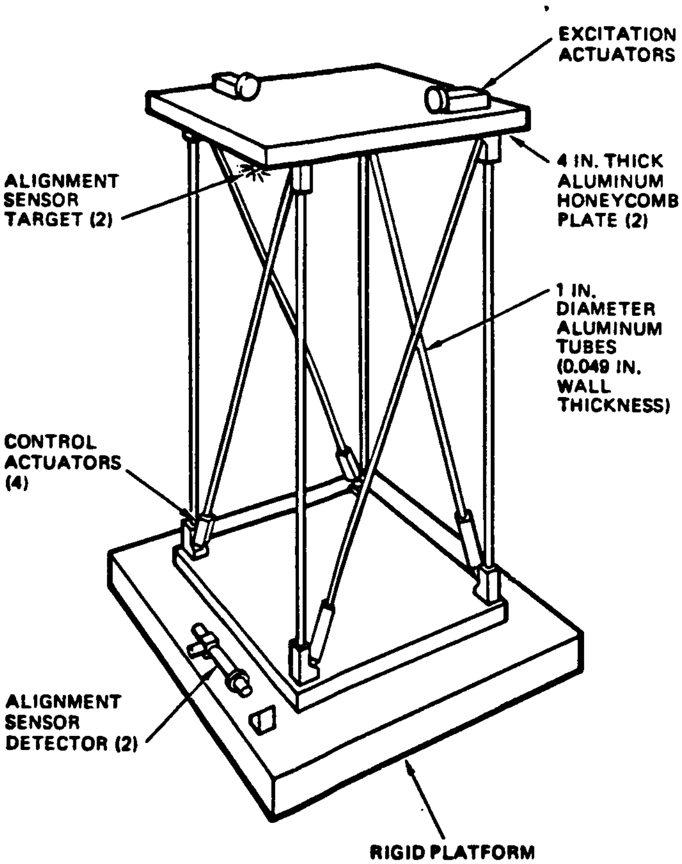


Figure 1: Large Space Structure Truss Experiment Configuration

displacements at each of two Surface Accuracy Measurement Sensors (SAMS) were recorded from each of these events for a total of twenty observations. The original sampling rate was 200 samples per second* which implies a Nyquist frequency of 100 Hz. To improve the ability of the Prony algorithm to resolve low frequency modes, which are around 1 Hz, the sampling rate was reduced to 40 sps. The data were pre–filtered to prevent aliasing using a five point Hanning smoothing algorithm developed at TRW. This algorithm preserves low frequency content and initial conditions while preventing phase shifts.

The Prony fits were employed only on the strong motion portion of the response time histories, up to 5.2 seconds. The goodness of fit was calibrated using a root mean square error norm between the Prony estimate and the actual data for each measurement. This error norm was used to assign uncertainty estimates to each of the measurements used in the Bayesian estimation procedure. Uncertainty estimates on physical parameters were chosen heuristically. The general rule was to assign large uncertainties (low weights) to prior parameter estimates in order to allow the model to match the test data as closely as possible.

* (sps)

## 3 Damping Models

The test data for all four sensors in one of the events designed to excite a lateral mode are shown in Figure 2. Also shown are the results of the Prony fit to the test data. It is readily apparent that exponentially damped sinusoids provide a good fit to the measured responses for moderately large motions. Hence a linear viscous damping model is valid in this regime. For small motions the response is seen to decay rapidly; indeed, it terminates entirely after 8 seconds of response. This suggests that Coulomb friction predominates in the small motion regime. (Recall that a Coulomb friction model results in a linear decay envelope.) The "grabbing" of the actuators suggests a transition from a dynamic coefficient of friction to a larger static coefficient as relative velocity in the actuators becomes small. A Dahl fiction model [1] was considered but rejected. The Dahl model provides a hyperbolic decay envelope which implies diminishing damping forces as response becomes small, contrary to our observations.

Modal damping ratios were assumed to be given by a superposition of intrinsic structural damping and viscous damping in the actuators, as expressed below for a given mode n:

$$\varsigma_n = \varsigma_{struct} + \Phi_n^T \mathbf{C} \Phi_n / (2\omega_n)$$

The voice coil actuators used for active control employ powerful cobalt-samarium magnets surrounding copper coils. Using energy principles one can derive the actuators' damping constant from their electro-magnetic properties. The result is

$$c = c_{EMF} + c_{eddy} = K_B^2/R \quad + \quad k_o B^2 V/\rho$$

The damping term due to back electro-motive force (EMF) is .269 lb/in/sec. The back EMF constant $K_B$ and resistance $R$ were supplied by the manufacturer and are 1.7 volts/foot/second and .66 ohms, respectively. The resistance of the control circuit power amplifiers was assumed to be negligible. The damping term due to eddy current or magnetic hysteresis was not known a priori since the magnetic field strength, $B$, and the volume of conductive material within the field, $V$, were unknown. The other variables are material resistivity $\rho$ and a constant $k_o$. Note that both forms of magnetic damping are proportional to the square of the magnetic field and inversely proportional to resistance. Their effect can be profound when the magnetic field is large and resistance is small. Since the eddy-current portion of damping was not known it was assumed to equal the back EMF portion in the prior model used for estimation.

An excellent fit to the test data was obtained with a reasonable set of physical parameters. The sequence of events for a typical sensor is shown in Figure 3. One can see that the Prony fit to the test data is quite good, and that the model's response predictions were greatly

[1] Philip R. Dahl, "Solid Friction Damping of Mechanical Vibrations", *AIAA Journal*, Vol. 14, No. 12, December 1976.

improved by the estimation procedure. Only three finite element analyses were required for the identification procedure to converge. Each design iteration took about five minutes of CPU time on an IBM 3084. Following is a list of the design variables employed.

*Column Stiffness* The effects of the end fittings on column stiffness were unknown. The thickness of a short thin-walled tube element at the top and bottom of the column was chosen to model this effect.

*Plate Mass* The honeycomb top plate contains a large, unknown amount of adhesive and the mass of the corner fittings is also unknown. The center plate thickness and a corner plate thickness were chosen to model total mass and torsional inertia. The model of the plate was stiffened by a set of rigid elements to allow only mass properties to be estimated.

*Structural Damping* The overall level of material damping was estimated. A lower bound of $\varsigma_{struct} = .1$ % was enforced. (An advantage of the structural optimization approach to parameter ID is that reasonable bounds can be placed on parameters.)

*Actuator Damping* A linear viscous damping constant representing the sum of all damping effects in the actuators was chosen.

*Actuator Gain* The amount of force delivered by each actuator in each event was estimated. Due to the effects of stiction, this varied from event to event.

Initial and final values of the physical model parameters along with their standard deviations are given in Table 1. The large increase in corner plate thickness makes it clear that a large proportion of the mass is in the column fittings. The column stiffness was increased somewhat to reflect the rigidity of these end fittings.

Table 1: Initial and Final Physical Parameters

| Parameter | | Value | | Standard Deviation | | |
|---|---|---|---|---|---|---|
| Description | Units | Initial | Final | Initial | Final | % Improvement |
| Tube t | inches | .049 | .105 | .2 | .066 | 67 |
| Corner Plate t | inches | 4. | 16.7 | 15. | 4.78 | 68 |
| Center Plate t | inches | 4. | .594 | 8. | .35 | 96 |
| Material Damping | % | .5 | .1 | 5. | 4.53 | 9.4 |
| Actuator Damping | lb/in/sec | .54 | 1.14 | 5. | .805 | 84 |

Initial and final modal parameters along with their uncertainties are given in Table 2. (Uncertainty in modal damping was not computed). A large amount of modal damping was provided by the actuators. Modal damping coefficients on the order of six percent in the lateral modes and eight percent in the torsional modes were obtained.

An advantage of the Bayesian statistical approach to parameter identification is the availability of variance or uncertainty estimates on dependent variables such as weight and frequency. The algorithm directly identifies physical parameters such as element stiffness, mass and damping. The behavior variables are found indirectly as a result of the model predictions. As a result, uncertainty in model parameters can be cascaded through the analysis to provide uncertainty in model predictions. For example, the lateral mode frequency was estimated to be .704 Hz $\pm$.0045 Hz, and the torsional mode was not estimated quite as well, being .992 Hz $\pm$.016 Hz. It is reassuring that the modal parameters predicted by the estimated model are less uncertain than the physical model parameters themselves. One can see from the test data that the natural frequencies are well known. A large uncertainty in an estimated physical parameter will not be manifested in large behavior uncertainties if the sensitivities to that parameter are small.

Table 2: Initial and Final Behavior Variables

| Parameter Description | Units | Value Initial | Value Final | Final Standard Deviation |
|---|---|---|---|---|
| Total Weight | lb | 389. | 328. | 60.1 |
| Lateral Mode f | Hertz | .614 | .704 | .0045 |
| Torsional Mode f | Hertz | 1.044 | .992 | .016 |
| Lateral Mode $\varsigma$ | % | 3.77 | 6.21 | |
| Torsional Mode $\varsigma$ | % | 5.42 | 7.70 | |

## 4 Conclusions

This relatively simple structure has demonstrated the validity of the parameter identification procedure for small to moderately sized structures. The procedure was found to be computationally efficient with the exception of mode shape sensitivity to model parameters, which consumed ninety percent of the entire computing budget. Scaling up to large models will require a much more efficient eigenvector derivative algorithm or the selective elimination of these computations for all but the most significant modes.

The use of physical parameter identification requires a great deal of thought in the selection of parameters which are both uncertain and have a large effect on response. This was found to be an iterative process between the analyst and the identification software. Initially a large set of parameters was chosen. This set was pruned down greatly as it was found that most parameters were not important or unidentifiable. The percent improvement in parameter uncertainty was very beneficial in this process. It was found that as parameters were removed from the estimation set, confidence in the remaining parameters increased. The choice of a minimum number off parameters is thus important. This process can take considerable analyst and computer time even for simple models.

The simplicity of the structure led to the lack of significant modes to identify. This procedure was exacerbated by the use of displacement sensors only. Accelerometers would measure response of high frequency modes with much greater resolution. It would be instructive to try this procedure on a more complex structure with accelerometers in addition to, or in lieu of displacement sensors.

The large amount of damping due to magnetic effects in the control actuators was a surprise. The use of this damping to augment an active vibration suppression system outside its computational bandwidth is a promising concept.

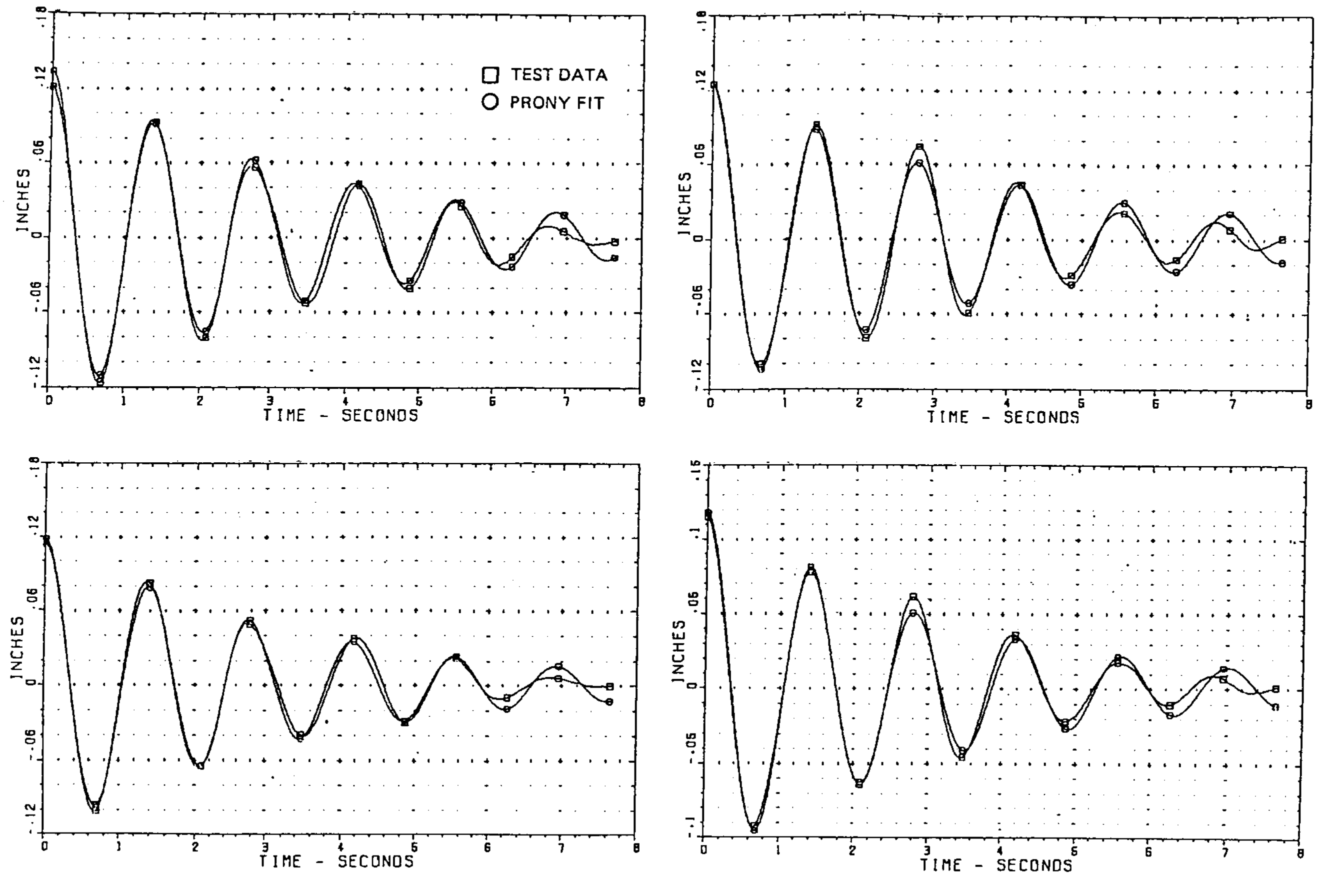


Figure 2: Prony Fit vs. Test Data for All Four Actuators in a Typical Load Event

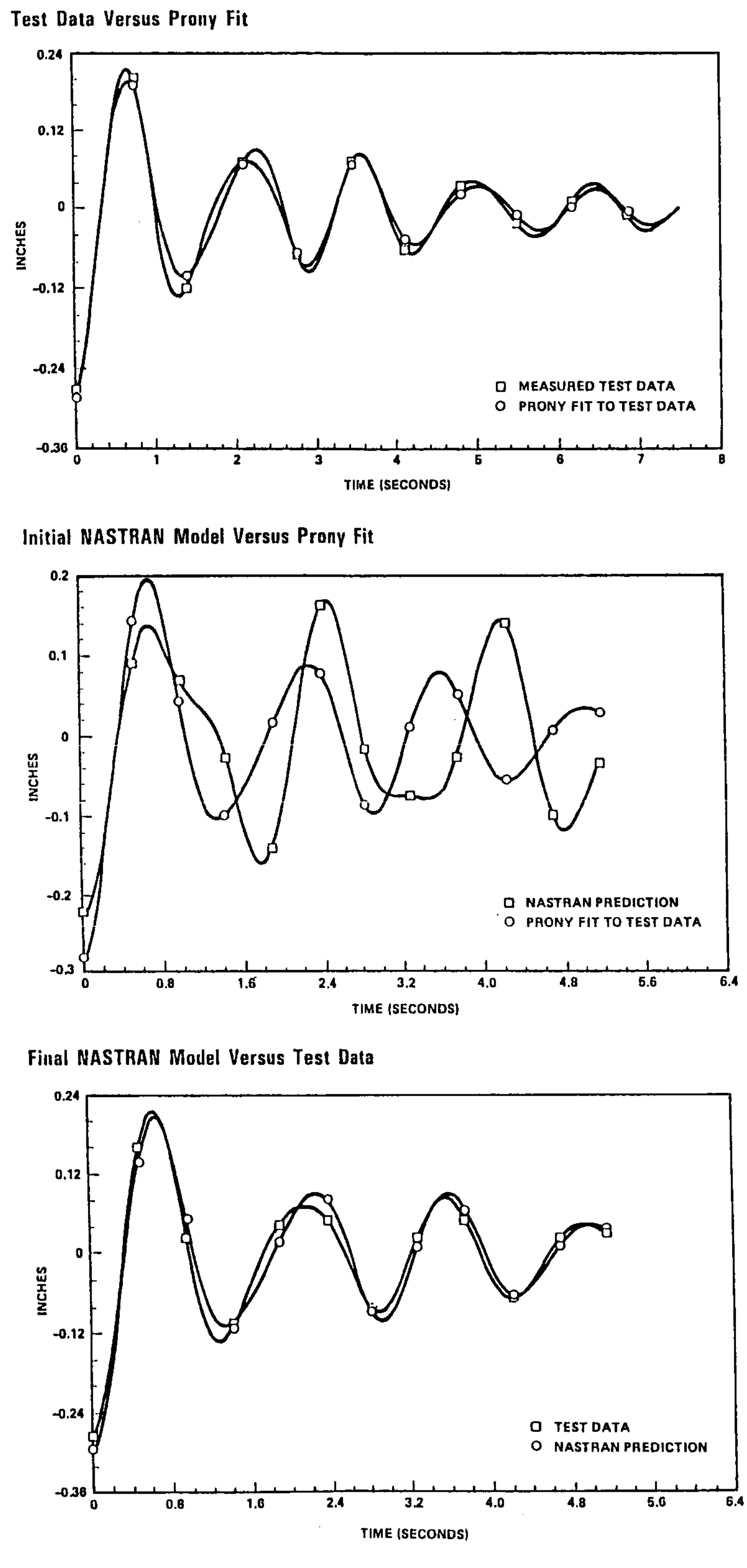


Figure 3: Identification Sequence for a Typical Sensor